

Simulation, construction and testing of a two-cylinder solar Stirling engine powered by a flat-plate solar collector without regenerator

Ali Reza Tavakolpour^{a,*}, Ali Zomorodian^a, Ali Akbar Golneshan^b

^a*Department of Mechanics of Farm Machinery Engineering, Shiraz University, Shiraz, Iran*

^b*Department of Mechanical Engineering, Shiraz University, Shiraz, Iran*

Received 7 October 2006; accepted 3 March 2007

Available online 24 April 2007

Abstract

In this research, a gamma-type, low-temperature differential (LTD) solar Stirling engine with two cylinders was modeled, constructed and primarily tested. A flat-plate solar collector was employed as an in-built heat source, thus the system design was based on a temperature difference of 80 °C. The principles of thermodynamics as well as Schmidt theory were adapted to use for modeling the engine. To simulate the system some computer programs were written to analyze the models and the optimized parameters of the engine design were determined. The optimized compression ratio was computed to be 12.5 for solar application according to the mean collector temperature of 100 °C and sink temperature of 20 °C. The corresponding theoretical efficiency of the engine for the mentioned designed parameters was calculated to be 0.012 for zero regenerator efficiency. Proposed engine dimensions are as follows: power piston stroke 0.044 m, power piston diameter 0.13 m, displacer stroke 0.055 m and the displacer diameter 0.41 m. Finally, the engine was tested. The results indicated that at mean collector temperature of 110 °C and sink temperature of 25 °C, the engine produced a maximum brake power of 0.27 W at 14 rpm. The mean engine speed was about 30 rpm at solar radiation intensity of 900 W/m² and without load. The indicated power was computed to be 1.2 W at 30 rpm.

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Keywords: Solar energy; Stirling engine; Flat-plate solar collector

1. Introduction

In view of energy management, solar energy is one of the most important renewable and non-depletable energy sources for generating power. There are several methods for converting solar heat into mechanical energy. One of these methods that theoretically associated with maximum efficiency is Stirling engine (or hot air engine). Stirling engine is a simple kind of external combustion engine. It is an old concept first proposed by Robert Stirling in 1816 (UK, patent no. 4081). Engines based upon his invention were built in many forms and sizes. These engines offer the possibly of a high-efficiency engine with lower exhausted emissions in comparison with the internal combustion engine. Hot air engines are clean and efficient and ran

almost silently on any combustible material such as field waste and biomass.

In 1970 and 1980s a huge amount of researches were conducted on Stirling engine for automobiles by companies such as General Motors and Ford. The main drawback is, the Stirling engine tends to run at a constant power setting that is not proper for automobiles. But this characteristic is perfect for applications such as water pumping. Studies about high-temperature Stirling engines have been extensively reported.

In most models, the engines operate with a heater and cooler temperature about 923 and 338 K, respectively [1].

The thermal limit for the operation of high-temperature Stirling engines depends on the material used for its construction. Engine efficiency ranges from about 30–40% resulting in a typical temperature range of 923–1073 K, and normal operating speed range is from 2000 to 4000 rpm [2]. On the other hand, the low-temperature differential (LTD) Stirling engine is a type of Stirling engine that can operate

*Corresponding author. Tel.: +989173147706.

E-mail address: alitavakolpour@yahoo.com (A.R. Tavakolpour).

Nomenclature

h	convective heat transfer coefficient
P	instantaneous pressure
m	gas mass
k	constant
V	volume
T	temperature
Q	heat
r	gas constant
S	exchanger plate area
W	work
x	working piston position
y	displacer position
Δ	difference
γ	gas heat capacity ratio
η	efficiency
ν	frequency
σ_x	dead space volume ratio = (V_{xds}/V_d)
ξ	compression ratio = $(V_d/V_p) = (S_d y_0)/(S_p x_0)$
τ	internal temperature ratio = (T_h/T_c)

φ	phase angle
α	displacer crank angle
β	power piston crank angle = $(\varphi - \alpha)$

Subscripts and superscripts

C	cold outer space/plate
c	cold inner space
0	reference parameter
d	displacer
ds	dead space
H	hot outer space/plate
h	hot inner space
loss	heat loss
p	working piston
reg	regenerator
*	dimensionless
**	dimensionless
'	isothermal process
pt	point

with relatively small temperature difference between hot and cold ends of the displacer cylinder [3]. The efficiency of a LTD Stirling engine is low in comparison with the high-temperature Stirling engines but it can be acceptable due to the availability of many low-temperature heat sources including solar energy which is inexpensive and safe. LTD engines may be of two designs. The first one uses single-crank operation where only the power piston is connected to the flywheel, called Ringbom engine. A short, large-diameter displacer rod in a precise-machined fitted guide has been used to replace the displacer connecting rod [3]. The other design is called a kinematic engine, where both the displacer and the power piston are connected to the flywheel through the cranks of the crankshaft [3]. Large amount of studies on LTD Stirling engines have been done during the recent century. Some of them are as follows:

Kolin [4] tested a number of LTD Stirling engines, over a period of many years. In 1983, he presented a model that worked on a temperature difference between the hot and cold ends of the displacer cylinder as low as 15 °C.

Senft [5] made an in-depth study of the Ringbom engine and its derivatives, including the LTD engine. His research in LTD Stirling engines resulted in a most interesting engine which had an ultra-low temperature difference of 0.5 °C. It seems difficult to create any advancement better than this result. Senft's work in 1991 [6], showing the principle motivation for using Stirling and general heat engines, developed an engine operating with a temperature difference of 2 °C or lower.

In 1993, Senft [4] described the design and testing of a small LTD Ringbom Stirling engine powered by a 60° conical reflector. He reported that the tested 60° conical

reflector, producing hot end temperature of 93 °C under running conditions, worked very well.

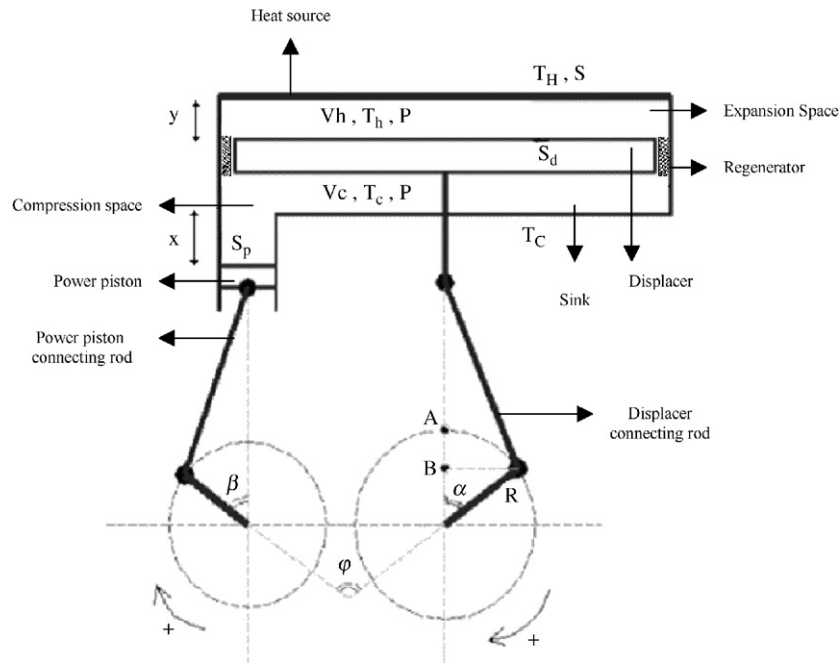
In 1997, Iwamoto et al. [7] compared the performance of a LTD Stirling engine with a high-temperature differential Stirling engine. Finally, they concluded that the LTD Stirling engine efficiency at its rated speed was approximately 50% of the Carnot efficiency.

In 2005, Kongtragool and Wongwises [8] investigated theoretically the power output of the gamma-configuration LTD Stirling engine. In this work, the power calculation was studied and discussed.

In 2005, Kongtragool and Wongwises [9] presented the optimum absorber temperature of a once reflecting full-conical reflector for a LTD Stirling engine and a mathematical model for the overall efficiency of a solar-powered Stirling engine was developed.

In 2006, Kongtragool and Wongwises [10] designed and constructed two single-acting, twin-power piston and four-power piston gamma-configuration LTD Stirling engines with heater temperature of about 589–771 K. The tested engines were equipped with regenerator to increase the thermal efficiency.

Solar energy is a proper heat source for LTD Stirling engine. Solar collectors as a special kind of heat exchangers are usually employed to convert the solar radiation into thermal energy. If a hot air engine is equipped with a concentrated solar collector as a heat source, the working temperature would be high and heat efficiency would be effectively improved. But this system always has to trace the direction of sun in the sky. The tracer is a complex system and can waste part of the energy. Flat-plate solar collectors need no tracer system and can absorb diffused solar energy radiation as well. As



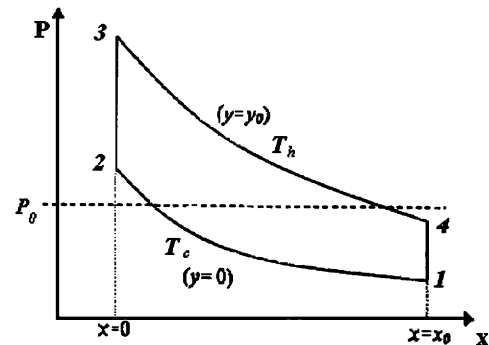
described above there is no published paper on experimental results of the kinematic LTD solar Stirling engine powered by a flat-plate solar collector without regenerator. In this research, the simulation, construction and testing of a solar LTD Stirling engine with flat plate solar collector as an in-built heat source and without application of the regenerator were investigated.

2. Mathematical model

A schematic sketch of a general LTD Stirling engine with gamma configuration is shown in Fig. 1. The Stirling cycle is composed of two isothermal and two isochoric processes (Fig. 2). The procedures of finite dimension thermodynamics were employed to calculate the optimized compression ratio according to the guidelines of Pierre Rochelle [11]. An optimization process, with work or efficiency as the objective function, is investigated. In the following description, the classical Schmidt assumption will be used except for the finite heat transfer between the working gas and the exchanger plates.

The gas mass content inside the engine is expressed by the sum of gas mass contents in the expansion and compression spaces together with the gas mass inside the regenerator:

$$\begin{aligned}
m &= m_{\text{c}} + m_{\text{h}} + m_{\text{reg}} \\
&= \frac{P}{rT_{\text{c}}} [xS_{\text{p}} + (y_0 - y)S_{\text{d}} + V_{\text{cds}}] \\
&\quad + \frac{P}{rT_{\text{h}}} [yS_{\text{d}} + V_{\text{hds}}], \tag{1}
\end{aligned}$$



where y_0 is the complete stroke of the displacer piston, P is the instantaneous pressure, m_{reg} is the mass of the gas inside the regenerator, V_{hds} and V_{cds} are the dead space volumes on the hot and cold sides. Since this research has been devoted to design a LTD solar Stirling engine without regenerator, thus m_{reg} was assumed to be zero in Eq. (1). Moreover, in the LTD Stirling engine, if a regenerator had been employed for improving the engine efficiency, the value of m_{reg} would have been negligible compared to $(m_c + m_h)$. This can be easily concluded from Fig. 1 according to the large volume of expansion and compression spaces in comparison with the small unavoidable dead volume of the regenerator.

To reduce the number of variables involved in the problem, dimensionless descriptions are defined as follows [11]:

$$P^* = \frac{P}{P_0}, \quad (2)$$

$$x^* = \frac{x}{x_0}, \quad (3)$$

$$y^* = \frac{y}{y_0}, \quad (4)$$

where x_0 is the complete stroke of the power piston. The reference mass m_0 and the dimensionless mass m^* are defined as follows:

$$m_0 = \frac{P_0 V_p}{r T_c}, \quad (5)$$

$$m^* = \frac{m}{m_0} = P^* \left\{ [x^* + (1 - y^*)\zeta + \zeta\sigma_c] + \frac{1}{\tau} [y^*\zeta + \zeta\sigma_h] \right\}. \quad (6)$$

Assuming the same dead space volumes $\sigma = \sigma_h = \sigma_c$ and letting $A = \zeta(1 + 2\sigma)$ and $B = \zeta((1/\tau) - 1)$. Substituting A , B and σ in Eq. (6):

$$m^* = P^*[x^* + A + B(y^* + \sigma)]. \quad (7)$$

Other dimensionless descriptions are as follows:

$$P^*(x^*, y^*) = \frac{m^*}{x^* + A + B(y^* + \sigma)}, \quad (8)$$

$$m_h^*(x^*, y^*) = \frac{m_h}{m_0} = m^* \left[\frac{\zeta(y^* + \sigma)}{\tau[x^* + A + B(y^* + \sigma)]} \right], \quad (9)$$

$$m_{h\max}^* = m_h^*(0, 1) = m^* \frac{\zeta(1 + \sigma)}{\tau[A + B(1 + \sigma)]}, \quad (10)$$

$$m_{h\min}^* = m_h^*(1, 0) = m^* \frac{\zeta\sigma}{\tau[1 + A + B\sigma]}. \quad (11)$$

The dimensionless mass transfer Δm^* between the volumes is defined as

$$\begin{aligned} \Delta m^* &= m_{h\max}^* - m_{h\min}^* \\ &= m^* \frac{\zeta}{\tau} \frac{(1 + A + \sigma)}{[A + B(1 + \sigma)][1 + A + B\sigma]}. \end{aligned} \quad (12)$$

To get the energy balance of the cycle, the dimensionless work W^* and transferred heats during the isothermal processes are expressed as

$$W^* = \frac{W}{P_0 V_p} = -\frac{1}{P_0 S_p x_0} \oint P S_p dx = -\oint P^* dx^*, \quad (13)$$

$$Q_c^* = \frac{1}{P_0 V_p} \oint P dV_c = \oint P^*(dx^* - \zeta dy^*), \quad (14)$$

where $V_c = xS_p + (y_0 - y)S_d + V_{cds}$.

$$Q_h^* = \frac{1}{P_0 V_p} \oint P dV_h = \zeta \oint P^* dy^*, \quad (15)$$

where $V_h = yS_d + V_{hds}$.

Solving Eqs. (13)–(15), we obtain

$$\begin{aligned} W^* &= -\oint P^* dx^* \\ &= -\left[\int_{pt1}^{pt2} P^*(x^*, 0) dx^* + \int_{pt3}^{pt4} P^*(x^*, 1) dx^* \right] \\ &= -m^* \left\{ \int_1^0 \frac{dx^*}{x^* + A + B\sigma} + \int_0^1 \frac{dx^*}{x^* + A + B(1 + \sigma)} \right\} \\ &= -m^* \ln(Z), \end{aligned} \quad (16)$$

where

$$Z = Z(\zeta, \sigma, \tau) = \left\{ \frac{1 + (1/A + B(\sigma + 1))}{1 + (1/A + B\sigma)} \right\}, \quad (17)$$

$$Q_c^* = -m^* \frac{\zeta}{B} \ln(Z), \quad (18)$$

To calculate the total heat absorbed by the working gas at the heat source, the heat lost in the regenerator is added to the transferred heat during isothermal process [11]:

$$\begin{aligned} Q_{reg} &= \Delta m \frac{r}{\gamma - 1} (T_h - T_c) \\ &= P_0 V_p \frac{\Delta m}{m_0} \frac{1}{\gamma - 1} (\tau - 1), \end{aligned} \quad (19)$$

$$Q_{loss} = Q_{reg}(1 - \eta_{reg}), \quad (20)$$

$$Q_{loss}^* = \frac{Q_{loss}}{P_0 V_p} = \Delta m^* (\tau - 1) \frac{1 - \eta_{reg}}{\gamma - 1} = -m^* G k_{reg}, \quad (21)$$

where

$$k_{reg} = \frac{1 - \eta_{reg}}{\gamma - 1}$$

and

$$G = G(\zeta, \sigma, \tau) = \frac{B(1 + A + \sigma)}{[A + B(1 + \sigma)][1 + A + B\sigma]}.$$

The total heat absorbed by the working gas at the hot source and the total heat rejected to the cold sink are as follows:

$$\begin{aligned} Q_h^* &= Q_h^* + Q_{loss}^* \\ &= \frac{-m^*}{((1/\tau) - 1)} \ln(Z) - m^* G k_{reg} \\ &= m^* \left[\frac{-\ln(Z)}{(1/\tau) - 1} - G k_{reg} \right], \end{aligned} \quad (22)$$

$$\begin{aligned} Q_c^* &= Q_c^* - Q_{loss}^* \\ &= \frac{m^* \ln(Z)}{\tau((1/\tau) - 1)} + m^* G k_{reg} \\ &= m^* \left[\frac{\ln(Z)}{\tau((1/\tau) - 1)} + G k_{reg} \right]. \end{aligned} \quad (23)$$

Finally, the efficiency of the engine can be calculated as

$$\eta = \frac{|W^*|}{Q_h^*} = \frac{-\ln(Z)}{[(\tau \ln(Z)/(1-\tau)) + k_{\text{reg}} G]} = \frac{\tau - 1}{\tau + k_{\text{reg}}(1-\tau)(G/\ln(Z))}. \quad (24)$$

The internal working gas temperature at expansion space is less than the temperature of the solar absorber and the air temperature inside the compression space is more than the cold sink. To calculate the real temperature of the working fluid inside the cylinder, heat transfer equations are used as follows [11]:

$$Q_c = \frac{hS_d}{v}(T_c - T_c) = \frac{hS_d}{v} T_c \left(1 - \frac{T_c}{T_c}\right), \quad (25)$$

$$Q_c^* = \frac{Q_c}{P_0 V_p} = \frac{hS_d T_H T_c}{v P_0 V_p T_H} \left(1 - \frac{T_c}{T_c}\right) = N \xi \frac{T_c}{T_H} \left(1 - \frac{T_c}{T_c}\right), \quad (26)$$

where

$$N = \frac{hS_d T_H}{v P_0 S_d y_0} = \frac{h T_H}{v P_0 y_0},$$

$$Q_h = \frac{hS_d}{v}(T_H - T_h) = \frac{hS_d}{v} T_H \left(1 - \frac{T_h}{T_H}\right), \quad (27)$$

$$Q_h^* = \frac{Q_h}{P_0 V_p} = \frac{hS_d}{v P_0 V_p} (T_H - T_h) = N \xi \left(1 - \frac{T_h}{T_H}\right). \quad (28)$$

T_c can be determined from Eqs. (23) and (26):

$$T_c = T_c - m^* \frac{T_H}{N \xi} \left[\frac{\ln(Z)}{\tau((1/\tau) - 1)} + G k_{\text{reg}} \right] = T_H \left\{ \frac{T_c}{T_H} - m^* \frac{\ln(Z)}{NB} \left[\frac{1}{\tau} + \left(\frac{1}{\tau} - 1\right) k_{\text{reg}} \frac{G}{\ln(Z)} \right] \right\}. \quad (29)$$

T_h can be determined from Eqs. (22) and (28):

$$T_h = T_H - m^* \frac{T_H}{N \xi} \left[\frac{-\ln(Z)}{(1/\tau) - 1} - k_{\text{reg}} G \right] = T_H \left\{ 1 - m^* \frac{\ln(Z)}{NB} \left[-1 - k_{\text{reg}} \left(\frac{1}{\tau} - 1\right) \frac{G}{\ln(Z)} \right] \right\}. \quad (30)$$

From Eqs. (29) and (30), the internal temperature ratio τ can be obtained:

$$\tau = \frac{T_h}{T_c} = \frac{1 - I[-1 - ((1/\tau) - 1)H]}{(T_c/T_H) - I[(1/\tau) + ((1/\tau) - 1)H]}, \quad (31)$$

where $H = H(k_{\text{reg}}, \xi, \tau, \sigma) = k_{\text{reg}}(G/\ln(Z))$ and $I = I(m^*, N, \xi, \sigma, \tau) = m^*(\ln(Z)/NB)$.

From Eq. (31):

$$\frac{1}{\tau} \left\{ \tau^2 \left[\frac{T_c}{T_H} + IH \right] - \tau[2I + 1] - IH \right\} = 0. \quad (32)$$

If T_c and T_h assumed to be close together approximately, then $\tau \approx 1$ and it results that $m^* \cong 1 + A$, $Z \cong 1 - (B/(A(1+A)))$, $\ln(Z) \cong -(B/(A(1+A)))$, $G \cong ((B(1+A+\sigma))/(A(1+A)))$, $H \cong -k_{\text{reg}}(1+A+\sigma)$, $I \cong -(m^*/N)(1/(A(1+A)))$ and $IH \cong m^*(k_{\text{reg}}/N)((1+A+\sigma)/(A(1+A)))$. Thus H and I are independent of τ [11]. The root of Eq. (32) is

$$\tau = \frac{(I + \frac{1}{2}) \left[1 + \sqrt{1 + [IH((T_c/T_H) + IH)]/(I + \frac{1}{2})^2} \right]}{(T_c/T_H) + IH}. \quad (33)$$

Reintroducing the real value of τ from Eq. (33) into the expression of W^* and η :

$$\eta = \frac{\tau - 1}{\tau + k_{\text{reg}}(1-\tau)(G/\ln(Z))} = \frac{\tau - 1}{\tau + (1-\tau)H} \cong \frac{\tau - 1}{\tau - (1-\tau)k_{\text{reg}}(1+A+\sigma)}, \quad (34)$$

$$|W^*| = m^* \ln(Z) \cong -m^* \frac{B}{A(1+A)} = \frac{(1 - (1/\tau))m^*}{(1 + 2\sigma)[1 + \xi(1 + 2\sigma)]}. \quad (35)$$

The dimensionless work W^* is reduced relative to the working piston swept volume but for the users a preferred reference volume is the overall volume, related to the engine bulk. This is equal to the working volume plus the displacer swept volume added to the dead space volumes. Then a new dimensionless work will be defined as [11]

$$|W^{**}| = \frac{|W|}{P_0(V_p + V_d + 2V_{ds})} = \frac{|W^*|}{1 + \xi(1 + 2\sigma)}. \quad (36)$$

2.1. Optimized compression ratio

The first optimization goal concerning LTD Stirling engines was to determine the optimum compression ratio ξ . For given values of the parameters in Table 1, the compression ratio was incremented to find the optimum value of ξ corresponding to maximum values of η and W^{**} . This was done by a computer program to determine the optimized compression ratio (Fig. 3).

In this research, an effort was devoted to design a solar Stirling engine powered by a flat plate solar collector

Table 1
Reference parameters

T_H (K)	T_c (K)	P_0 (pa)	σ	v (rev s ⁻¹)	h (W m ⁻² K ⁻¹)	y_0 (mm)	γ	η_{reg}
373	293	100 000	0.1	0.5	10	55	1.4	0

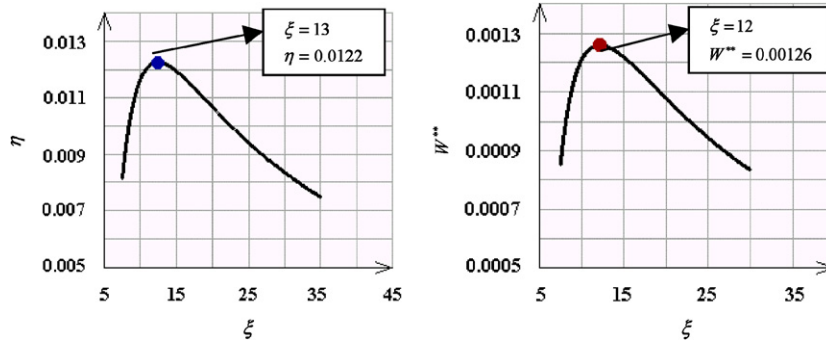


Fig. 3. Efficiency and dimensionless secondary work versus compression ratio.

without regenerator. The practical temperature of a flat-plate solar collector is around 100 °C or more. Therefore TH selected to be 100 °C and T_c assumed to be about the wet bulb temperature of the environment. Since the regeneration process was ignored in this study, $\eta_{reg} = 0$. The convective heat transfer coefficient h , assumed to be $10 \text{ W m}^{-2} \text{ K}^{-1}$ on both cold and hot plates [11].

It can be seen in Fig. 3, there are optimum values of the compression ratio. The optimized compression ratio assumed to be 12.5 to maximize the secondary work and efficiency.

2.2. Internal temperature of expansion space and compression space

Substituting the optimized compression ratio and parameters of Table 1 and Eq. (33) into Eqs. (29) and (30), the internal temperature of expansion and compression spaces were computed. A computer program was written to solve the equations. The corresponding values of T_h and T_c were calculated as follows:

$$T_h = 337 \text{ K}, \quad T_c = 328 \text{ K}.$$

2.3. Schmidt theory

The Schmidt theory is one of the isothermal calculation methods for Stirling engines. This theory provides for harmonic motion of the reciprocating elements but retains the major assumptions of isothermal compression and expansion and of perfect regeneration. It, thus, retains highly idealized, but is certainly more realistic than the ideal Stirling cycle [1]. The assumption of simple-harmonic volume variation permits pressure, P , to be expressed as a function of crank angle, α , and leads to closed form solutions for work per cycle [12]. The Schmidt formula may be shown in various forms depending on the notations used and can be arranged for gamma-configuration Stirling engines [4]. Since the internal temperatures of expansion and compression spaces, considering the proposed model, were calculated in Section 2.2, the Schmidt theory with all

its related assumptions can be employed to determine the theoretical output work and optimized phase angle of the engine. According to the mechanical configuration shown in Fig. 1, the Schmidt formula can be arranged as follows:

$$V_h = V_{hds} + \frac{V_d}{2}(1 - \cos(\alpha)), \quad (37)$$

$$V_c = V_{cds} + \frac{V_p}{2}(1 - \cos(\alpha - \varphi)) + \frac{V_d}{2}(1 + \cos(\alpha)), \quad (38)$$

$$\begin{aligned} m &= \frac{P}{r} \left[\frac{V_c}{T_c} + \frac{V_h}{T_h} \right] \\ &= \frac{PV_d}{2rT_c} \left[\frac{2\sigma_h}{\tau} + \frac{1}{\tau} - \frac{1}{\tau} \cos(\alpha) + 2\sigma_c \right. \\ &\quad \left. + \frac{1}{\xi} - \frac{1}{\xi} \cos(\alpha - \varphi) + \cos(\alpha) + 1 \right], \end{aligned} \quad (39)$$

$$P = \frac{2mrT_c}{V_d F(\alpha)}, \quad (40)$$

where $F(\alpha) = (2\sigma_h/\tau) + (1/\tau) - (1/\tau) \cos(\alpha) + 2\sigma_c + (1/\xi) - (1/\xi) \cos(\alpha - \varphi) + \cos(\alpha) + 1$.

$$W_h = \oint P dV_h = mrT_c \int_0^{2\pi} \frac{\sin(\alpha)}{F(\alpha)} d\alpha, \quad (41)$$

$$\begin{aligned} W_c &= \oint P dV_c \\ &= \left[\frac{mrT_c}{\xi} \int_0^{2\pi} \frac{\sin(\alpha - \varphi)}{F(\alpha)} d\alpha \right] - \left[mrT_c \int_0^{2\pi} \frac{\sin(\alpha)}{F(\alpha)} d\alpha \right], \end{aligned} \quad (42)$$

$$W_{total} = W_h + W_c = \frac{mrT_c}{\xi} \int_0^{2\pi} \frac{\sin(\alpha - \varphi)}{F(\alpha)} d\alpha. \quad (43)$$

W_{total} is the total work done per cycle. Using the results of Section 2.2 and the information of Table 1, the total work done per cycle and the optimized phase angle can be evaluated.

2.4. Total work done per cycle and optimized phase angle

A computer program was written to analyze the Eq. (43) and plot the total work done per cycle as a function of phase angle to calculate the optimum phase angle and total work (Fig. 4).

In this research, the engine was designed with two cylinders, so the total work done per cycle is $2 \times 1.18 = 2.36$ (J) theoretically without any regeneration process.

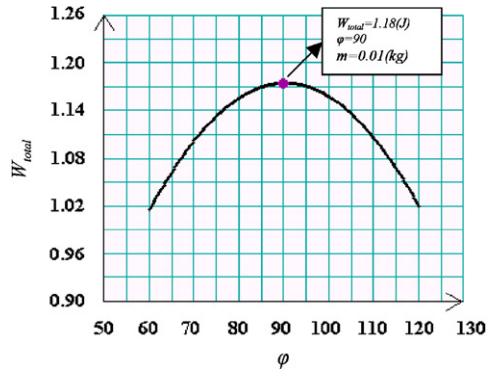


Fig. 4. Total work done per cycle versus phase angle.

Table 2
Design parameters

Displacer	
Bore $\times$ stroke (m)	0.41×0.055
Power piston	
Bore $\times$ stroke (m)	0.13×0.044
Phase angle	90°
Flat-plate collector dimensions (m)	1×0.5

3. Engine design and construction

Main engine design parameters are shown in Table 2 according to mathematical model. The schematic diagram of the engine was shown in Fig. 5. It is designed with two separate cylinders and Gamma-configuration arrangement.

The power cylinders were made of a steel pipe and the power pistons were made of an aluminum bar. The power pistons were turned to match the power cylinder bores. The clearance between the power piston and power cylinder was 0.01 mm $\times$ 15 mm thick Teflon pan was used to construct the power piston connecting rod. The solar absorber and the cold plate were made of 3 mm thick aluminum plates. The crank shaft was made of 12 mm thick steel shaft with eight Teflon cranks and crank pins. The crank shaft was supported by six self-align ball bearings that were located inside six wooden casing. The flywheel was attached to the middle part of the crankshaft. The flywheel was constructed from 15 mm thick Teflon pan with several 0.14 kg steel dead weights attached around it to decrease the speed fluctuations.

4. Measurement apparatus

The testing facilities are shown in Fig. 7. Since the engine speed was low, an accurate photo tachometer with ± 0.1 rpm accuracy was used to measure the engine speed. Cooling water was used in the cold sink. To measure the temperatures of the absorber and cold sink, some SMT-160 thermal sensors were attached to the aluminum plates of heat exchangers. The accuracy of the temperature measurement was $\pm 0.5^\circ\text{C}$. To have a wide range of collector temperature, a flat reflector was attached to the wooden

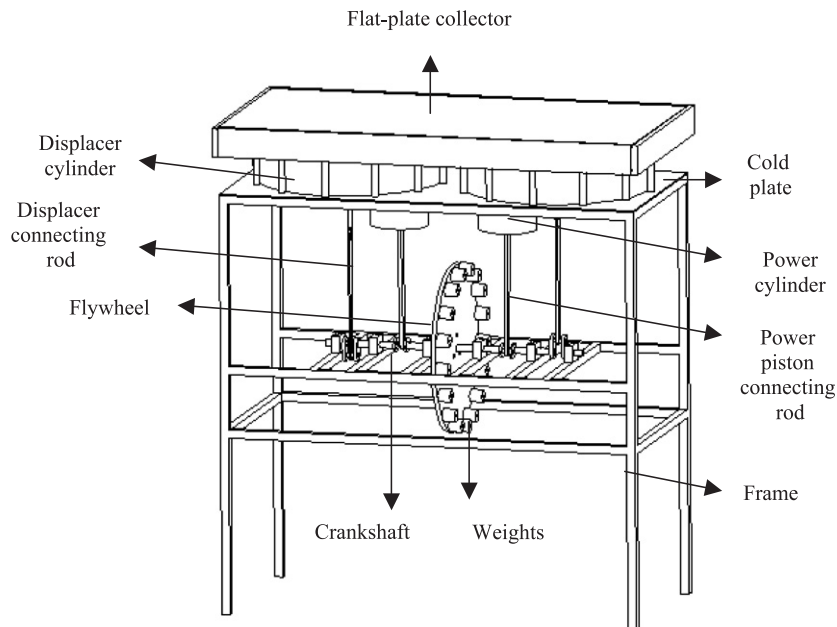


Fig. 5. Solid model of the LTD solar Stirling engine without regenerator.

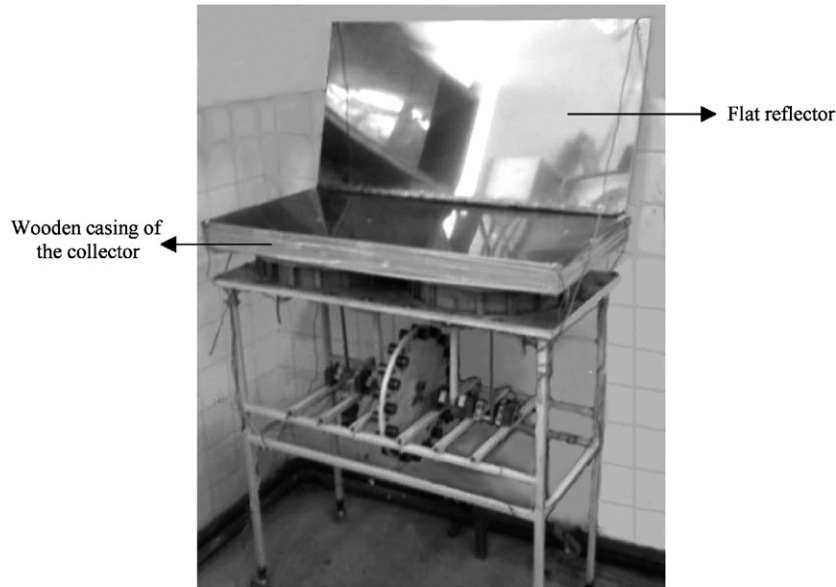


Fig. 6. View of the LTD solar Stirling engine without regenerator.

casing of the collector to enhance the solar energy intensity on the absorber plate of the collector (Fig. 6).

The solar intensity radiation was measured and recorded by a Casella solarimeter every 5 min ($0\text{--}2000\text{ W m}^{-2}$, $\pm 1\text{ mmv W m}^{-2}$).

Keeping in mind that generally the rope brake dynamometer with a spring balance and loading weights is used to measure the engine torque at different engine speed and the engine torque can be determined from the difference of spring balance reading and loading weight. Since the speed of the LTD solar Stirling engine is clearly low compared to the speed of internal combustion engines, an attempt has been made to employ another dynamic method for evaluating the torque and brake power of this low-speed LTD Stirling engine. Some weights, a rope and a brake drum have been used to measure the engine torque and the brake power (Fig. 7). The weights were hanged around the rotary brake drum attached to crankshaft through the rope and then, rotating the brake drum would cause the hanging weights to be elevated. Therefore, the hanging weights would act as a braking force against the rotary brake drum and the engine torque can be computed by multiplying the downward force of the loading weights by the brake drum radius once the constant engine speed is displayed on the digital tachometer. The brake drum diameter was 0.02 m and it was directly attached to the axis of the engine crankshaft through a central hole inside it.

5. Results and discussion

5.1. Indicated power

The mean indicated power can be calculated approximately from

$$P_{\text{Indicated}} = \frac{W_{\text{total}} n_{\text{mean}}}{60}, \quad (44)$$

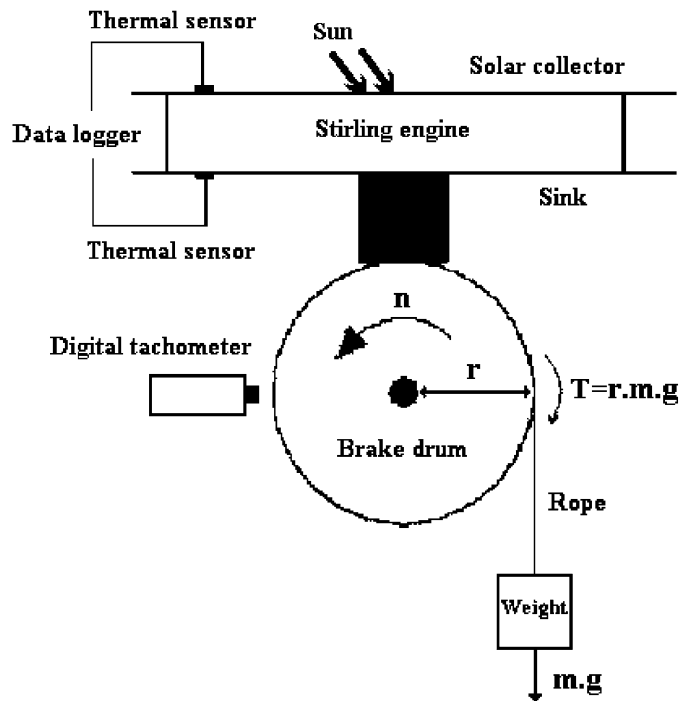


Fig. 7. Schematic diagram of the solar Stirling engine rig and testing facilities (r : brake drum radius, m : mass of the loading weights, T : engine torque, n : engine speed, g : gravitational acceleration).

where W_{total} is the total work done per cycle computed in Section 2.4. and n_{mean} is the mean engine speed at mean collector temperature of 110°C and sink temperature of 25°C . n_{mean} was measured to be about 30 rpm. The indicated power was calculated to be about 0.6 W for each cylinder and thus the total indicated power for two cylinders would be 1.2 W.

5.2. Engine performance

The engine was constructed and primarily tested at Shiraz University from 1 to 6 August 2005. Fig. 8 shows the engine torque variations versus the engine speed. In this figure, the engine torque T is calculated by multiplying the brake drum radius r by the downward braking force mg as shown in Fig. 7.

$$T = rmg, \quad (45)$$

where m is the mass of the loading weights and g is the gravitational acceleration. Fig. 9 shows the brake power versus engine speed. The brake power can be calculated from dynamics equations as [13]

$$P_{\text{Brake}} = \frac{2\pi Tn}{60}, \quad (46)$$

where n is engine speed in rpm and T is the engine torque in Nm.

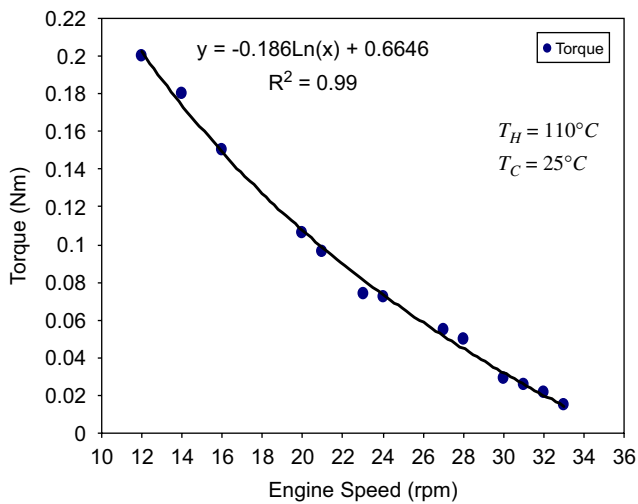


Fig. 8. Engine torque versus engine speed.

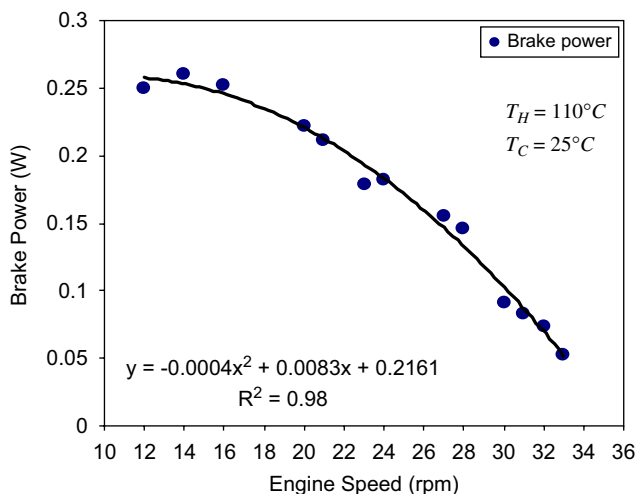


Fig. 9. Brake power versus engine speed.

According to Fig. 7, the load is gradually applied to the brake drum, and then the engine speed would be gradually reduced till a certain applied load would finally shut down the engine. The experiments were conducted during the time interval of 11.30–13.00 o'clock, when the mean collector temperature was measured about 110 °C with minimum fluctuations. The characteristics of torque and brake power variations represented in Figs. 8 and 9, are similar to the results of the research published by Kongtragool and Wongwises [10]. In these figures, it can be noted that the torque and the brake power decrease with an increase in engine speed. This reduction may be attributed to the lack of rather good heat transfer at higher engine speed as well as increasing in friction.

5.3. Relationship of collector temperature and engine speed at no-load condition

The no-load speed of the engine at various collector temperatures for two different sink temperatures is shown in Fig. 10.

It can be illustrated that, the engine speed has been increased with an increase in the collector temperature. It is also observed that with decreasing the sink temperature, the engine speed is effectively increased. These improvements may be due to the high heat transfer potential at high-temperature gradients.

5.4. The effect of regenerator application on the engine thermal efficiency

In this research, the engine was designed without regenerator. An important question would be raised about the amount of efficiency reduction due to exclusion of the regenerator. In order to figure out the effect of regenerator application, a computer program was written to analyze Eq. (34). The effect of regenerator efficiency variations

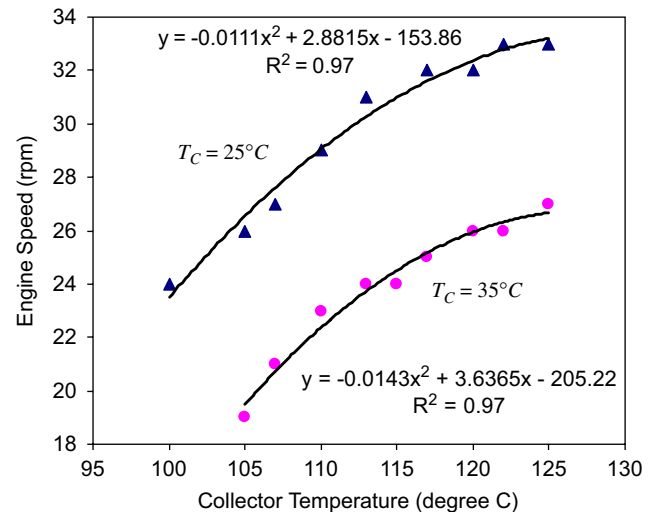


Fig. 10. No-load speed of the engine versus collector temperature at $T_C = 25$ and 35°C .

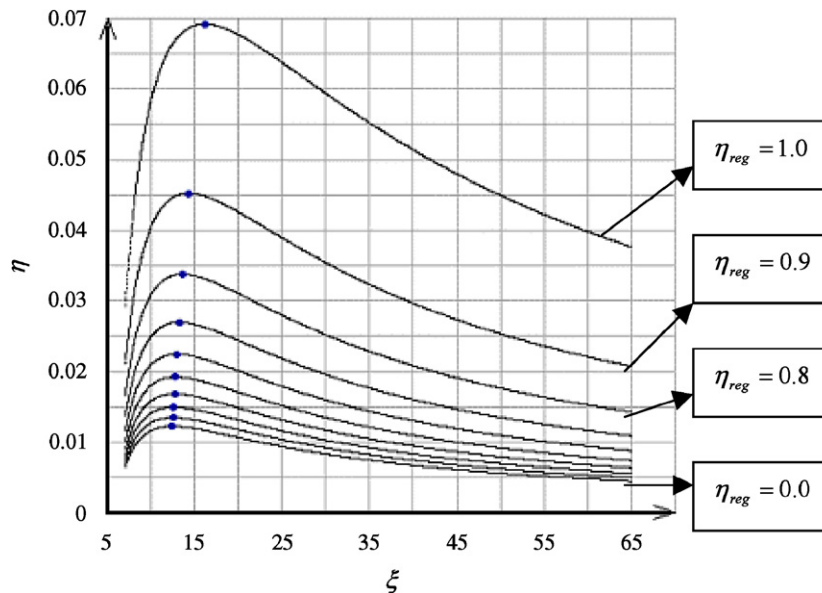


Fig. 11. Engine thermal efficiency versus compression ratio for different regenerator efficiencies.

versus the theoretical thermal efficiency of the engine was investigated. All reference parameters assumed to be constant according to Table 1 except the regenerator efficiency. The results were shown in Fig. 11.

It can be argued that the engine thermal efficiency was effectively improved for the regenerator efficiency of 1.0. Moreover, the engine thermal efficiency showed a diminishing trend towards the regenerator efficiency of zero. It was also observed that the optimized compression ratio shifted to right due to an increase in the efficiency of regenerators.

6. Conclusions

In this research, the possibility of generating power from low temperature sources such as flat-plate solar collectors was investigated. A very simple two-cylinder LTD solar Stirling engine without regenerator was designed and primarily tested. Although the regenerator was omitted to determine the minimum possible output power and to simplify the engine structure, but results were incredible.

The mean engine speed was measured to be 30 rpm at collector temperature about 107–113 °C and sink temperature of 25 °C which are approximately similar to the desirable reference parameters in Table 1. The procedure of finite dimension thermodynamics is suggested to predict the engine speed and the internal temperature of expansion and compression spaces. Therefore, the internal temperature ratio can be calculated and used in Schmidt theory to compute a more reliable indicated power. The indicated power was calculated to be 1.2 W at 30 rpm and the brake power was measured to be about 0.1 W at 30 rpm.

The application of an efficient regenerator is emphasized to increase the thermal efficiency. According to Fig. 11, for

the regenerator efficiency of 1.0 at ideal conditions the engine thermal efficiency is 0.069 whereas for the regenerator efficiency of zero, the engine thermal efficiency is about 0.0122. Therefore, by using an efficient regenerator, it is possible to increase the thermal efficiency six times more.

Upon optimization, the optimal compression ratio was computed to be 12.5 with collector temperature of about 100 °C, sink temperature of 20 °C and without any regeneration process. If an efficient regenerator was employed, the optimal compression ratio would be increased to 16 according to Fig. 11.

The corresponding theoretical efficiency of the engine for the mentioned designed parameters was calculated to be 0.012 for zero regenerator efficiency.

Although the classical Schmidt theory is considered as one of the most widely used method in designing the Stirling engines, but its predictions seem to have limitations due to some unreal assumptions of this theory such as finite heat transfer between the working gas and the heat exchanger plates. In this study, an effort was conducted to improve this drawback in Schmidt theory by calculating the more accurate gas temperature inside the exchangers. This was done by employing appropriate heat transfer equations together with the proposed thermodynamics method to calculate more realistic working gas temperatures. This finding results in a better estimation of indicated power of the LTD solar Stirling engine.

Acknowledgment

The authors wish to express their deep gratitude to Prof. James Senft from University of Wisconsin-River Falls for cooperating in this study.

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